

## TURBINE AND ROTOR THEREFOR

### FIELD OF INVENTION

This invention relates to a turbine and rotor therefor with an axis of rotation substantially parallel to a gas / fluid flow. More particularly, this invention relates to an un-enclosed wind / water turbine or a rotor / impellor housed within a duct extracting or converting energy from / to a moving stream of gas or fluid.

### BACKGROUND

On the most part, modern wind turbine rotors are of low solidity and have few long straight blades of "airfoil" section revolving around a central horizontal axis with a large proportion of their blade areas situated within the inner half of their diameters. The very high tip speed ratios involved in obtaining maximum efficiency can greatly add to noisy operating conditions of these turbines.

The present inventor has realised that the outer one third of any turbine rotor does most of the useful work in converting the kinetic energy from a moving gas / fluid flow into available torque as torque is a function of force x radius, and also that it is more beneficial in power production to have an increased gas / fluid flow velocity rather than having an increase in the overall size of the turbine rotor, the present invention seeks to situate the majority of the working surface being presented to the gas / fluid flow in this outer region in an effort to achieve a high mechanical efficiency within a design that remains relatively basic, un-encumbered, free-flowing and is does not rely on high tip speed ratios.

The maximum theoretical percentage of energy that can be extracted from a wind flow is % 59.3 (the BETZ limit) and this invention has shown University supervised wind tunnel test results supporting of a maximum co-efficient of power above % 52

### STATEMENT OF THE INVENTION

This invention seeks to provide a high efficiency output from any wind, water, steam, or gas turbines that have a rotation axis generally parallel to fluid / gas flow by using a rotor design that increases through-flow velocity by having a total flow outlet area formed by the gaps or voids between its blades / vanes much greater than the inlet flow area pre-determined by the maximum rotor diameter and also situating the majority of the working surface area in its outer extremities maximising total power output for its size, and comprises of a central hub or shaft rotatable about an axis substantially parallel to gas / fluid flow supporting a plurality of integral blade / vane units equi-distant and radially arranged around the said hub or shaft that each contains within, an integrally formed combination of typically short inner blade or "wing" portions extending substantially outwards from the said hub or shaft, and a substantially forwardly extending outer "vane" section preferably normal to and joined to the outer, forward extremity of the said inner portion with the whole vane / blade unit being mounted onto the hub / shaft so as to form a helix or pitch angle between its outer radial extremities and the said hub / shaft axis centre line preferably between 0 - 6 degrees more than the resultant angle corresponding to the resultant sum of the incoming gas / fluid flow vector and the tangential gas / fluid flow "headwind" component due to rotation and the complete revolving rotor assembly encouraging the flow from being generally parallel to its axis to moving outwardly and rearwardly in an increasing "helical" path preferably exiting in most part past the forwardly projecting outer vane sections that preferably contain within each, an appropriate angle of incidence at any section along their longitudinal axis, to the resultant fluid / gas vector flowing past that same specific section irrespective of where that section is located or its specific cross section and most preferably that angle of incidence is between 5 and 15 degrees.

As each blade / vane unit is preferably balanced both in weight distribution and twisting ( moment ) forces due to " lift " from fluid / gas flow , about its own central mounting point centerline normal to the hub or shaft , there is less or no need for an annular stiffening rim at its forwardmost perimeter or midsection connecting it to the other blade / vane unit /s , which simplifies manufacturability whilst retaining the ability to operate at much higher angular velocities as could otherwise have been expected without excessive flexing or failure due to large bending or twist stress levels and this design also retains the possibility of the inclusion of blade / vane articulation to a differing " attack " angle for the purpose of speed limiting or start - up situations .

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10 In the preferable form of this invention, slightly curved slots have been positioned in the outer rearward ends of the outer portions or vanes extending approximately perpendicular to the "resultant" flow in this region resembling in cross- section a slotted wing or "Fowler flap" greatly increasing the lift forces in this said region , enabling the rearmost end of the blade / vane unit to oppose and balance large moment forces formed by the use of extra ordinarily pronounced forward projecting vane portions and if desired , to completely 15 overcome these forces and twist the vane section to a lesser helix or "attack angle" feathering the vane further into the incoming flow enabling maximum speed limiting due to vane / blade flex at a pre- determined flow velocity .

20 Preferably , this turbine rotor design is a wind turbine , how ever this does not diminish its ability to provide a useful design alternative to any gas , fluid or steam turbines that may be employed in a variety of different situations where the maximum available space may be limited .

To assist with understanding the invention , reference will now be made to the accompanying drawings which show details of some examples of this invention however it is to be understood that the features illustrated in and described with reference to the drawings are not to be construed as limiting the scope of the invention .

5 In the the drawings :

Figure 1 shows a top view of the preferred embodiment .

Figure 2 shows the frontal view of the preferred embodiment , excepting the number of blades and # 2 being the direction of rotation in this instance.

10 Figure 3a , 3b and Figure 4 shows various section cutaway views of the preferred integral blade / vane unit depicting the relationship between the resultant gas / fluid flows 11 # and the the angles of incidence  $\alpha$ , to the blade / vane cross section at that same position with Figure 3 b also showing the preferred cross section in the vicinity of slot # 5

Figure 5 is an isometric view of the preferred embodiment.

15 Figure 6 " A " depicts a simplified multistage turbine embodiment having a second stage with a differing angle of attack on its blades and direction of rotation than the first stage to maximise resultant lift forces directed into torque at the hub and may have pre - rotor vanes.

Figure 6 " B " shows a method of achieving blade angle of attack adjustment with a mounting shaft protruding from the lower central mounting line # 8 of the blade / vane units which can be articulated by mechanical means built into the hub .

20 Figure 6 " C " shows an embodiment without slots that may be more easily manufacturable using the pressed metal or vacuum formed methods that also has an annular rim attached to the forwardmost perimeter of the blade / vane units to increase rigidity.

25 Figure 6 " D " depicts a simplified 2 stage fan that may be enclosed in a duct to suit air craft / hovercraft that has a second rotor with a differing rotation direction and blade pitch angle so as to enable the flow produced by rotation to exit at a typically lesser angle without sacrificing pitch length and so actual rearward thrust ( that consists of V1 axial plus V 2 axial components )

Figure 7 shows a face - on view of a blade / vane unit of the preferred embodiment with reference to the design formulae scaled for any given turbine diameter , with :-

" D " = maximum rotor diameter

CL max = the maximum lift co-efficient for a blade or wing unit area

Y = the total area of blade / vane rearwards of the central mounting point line # 8

X = the total area of blade / vane forward of the central mounting point line # 8

A in = area of flow intake ( = rotor radius squared x phi . )

A circ = area of flow exiting outwards at perimeter of vanes .

A thru = area of flow exiting at rear of rotor

$\Theta$  = pitch or attack angle of blade / vane units to the hub / shaft axis

$\omega$  = the angle between the vanes leading inner edge # 7 to the hub / shaft axis #6.

$\phi$  = the angle between the blade section leading edge to the central line # 8

# 1 = outer vane section

# 3 = inner blade section

# 6 = hub / shaft axis centerline

# 8 = the central line passing through the centroid of areas #10 perpendicular to the hub or shaft axis .

# 9 = the area of junction between the blade/ vane and the central hub or shaft

# 10 = the centroid of area about which the total sum areas of the blade and vane sections are considered to be centered upon .

# 11 = the " Resultant" flow vector comprising the sum of the axial flow velocity the radial flow velocity and the tangential flow velocity due to rotation .

# 12 = the incoming flow direction

# 14 = the angle between the vane / blade outer trailing edge and the hub shaft axis .

Figure 8 is the conclusional page to a University supervised wind tunnel test on a 765 mm Dia. rotor.

Referring to figure 1,  
A plurality of equi-spaced integrally formed "crooked" blade / vane units that consist of most preferably an inner airfoil section blade #3 that extends substantially outwards radially from a central hub or shaft #4 at a slight rearward angle, each inner blade having a leading edge also rearwardly sloping between 5 and 60 degrees from normal and a substantially forwardly protruding vane section #1 integrally formed with and joined to its outer forward edge and the whole blade / vane unit generally twisted in a helix or pitch angle  $\Theta$  about the hub/shaft central axis #6 that is preferably parallel to the gas / fluid flow direction #12 so as to maximise the lift or deflection forces obtained from the resultant flow and converted into available torque.  
The vane section #1 is preferable of an air foil cross section that diminishes in chord length in proportion to its distance away from the inner blade section #3 to form a curved outer point leading into the oncoming flow.

15 Referring to Figure 7  
The vanes #1 preferably contain slot/s #5 within their outer rear section that are set approximately normal to the resultant flow #11 past that same said section and may be curved, each slot being quite narrow with a smoothly rounded exit edges so as to direct a portion of the gas / fluid flow through to the rearward face of the vane / blade unit providing an increase of "lift" forces in this region (Figure 3 b) and most preferably forming a secondary "curved" or airfoil cross section in this area and enabling a large increase in the co-efficient of lift in this rearmost vane area being useful in balancing a pronounced front section of vane area having a lesser co-efficient of lift per unit area which can allow for equilibrium to be maintained due to moment or twisting forces about the central line #8 passing through the total vane / blade area centroid #10 normal to the hub axis #6 .  
Also, preferably, the total mass forward of the central line #8 (area x-x) is equal to the total mass rear of the central line #8 (area y-y) enabling a fully balanced blade design to be achieved as the central line #8 passes through the centroid of area #10 perpendicular to the hub axis centerline #6 .

20 The central hub #4 could be constructed in a variety of shapes and sizes but preferably has a diameter of between 0.2 and 0.4 of the total rotor diameter, increasing in diameter in a smoothly curved cone shape towards its rear helping to direct flow outwards and rearwards without imparting excessive turbulence and providing a possible housing for blade articulation mechanisms, a generating unit or connection to a suitable output shaft and / or support bearings.  
It can be seen from Figure 1 that the general shape of the complete rotor is designed to impart a fluid or gas flow pattern that has a substantial outward direction as it moves further into and completely through the rotor .

25 40 As the total exiting flow area "A thru" plus "A Circ" is much larger than the total inlet flow area "A In" with Volume - in being equal to Volume - out and Volume equal to velocity x area then it follows that from Bernoullies principle that there must be an increase in velocity inside / forward of the rotor or a pressure drop outside / rearward of the rotor all of which improving turbine rotor performance over prior art .  
45 All the leading edges are preferably suitably rounded to minimise turbulence and a good surface finish is applied to all sections with the inner blade section being of sufficient strength to adequately transform or direct the sum of the deflection and "lift" forces from the blades, vanes and slots due to fluid / gas flow into torque at the hub or shaft and to be able to withstand centrifugal and bending forces due to the total mass revolving at the maximum rated speed in extreme conditions .